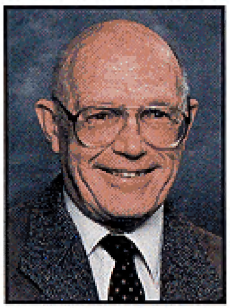
Shaft CenterLINES

A simple test for potential fluid-induced rotor instability



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Fluid-induced instability problems, such as oil whirl and oil whip, still often occur as an unpleasant surprise in the field, but machines are regularly purchased with no stability test.

By far, the best way to test for stability of a rotating machine is to run it at its regular operating speeds and conditions. While at these conditions, perturb the rotor in the forward direction (with rotative speed) by a forward circular force (or moment) of known amplitude and phase at frequencies from slow roll to well over rotative speed. Observe the amplitude and phase of the resultant rotor motion at each perturbation speed.

At each speed, the known input force vector at the perturbation speed is divided by the observed motion vector of that speed in order to yield the dynamic stiffness of the machine while at its usual operating conditions.

The general dynamic stiffness equations of a simple rotor system (ignoring gyroscopics and assuming symmetry) are:

$$K_{DIR} = K_O - (\omega - \lambda\Omega)^2 \bullet M_{FL} - \omega^2 M_R$$

$$K_{QUAD} = jD_B \bullet (\omega - \lambda\Omega) + jD_S\omega$$

where K_{DIR} is the Direct Dynamic Stiffness
K_{QUAD} is the Quadrature Dynamic Stiffness
K_O is the direct system stiffness
ω is the perturbation (or whirling) speed
Ω is the rotative speed
λ is the ratio of circumferential fluid whirling to rotative speed
M_{FL} is the fluidic inertial effect (not treated in this article)
M_R is the effective rotor mass
D_B is the bearing (or seal) damping
D_S is the system damping away from the bearing (or seal)
W is the total rotor weight

The Direct Dynamic Stiffness and the Quadrature Dynamic Stiffness graphs are plotted separately as a function of the perturbation speed. The typical results are shown in Figure 1.

Graphically:

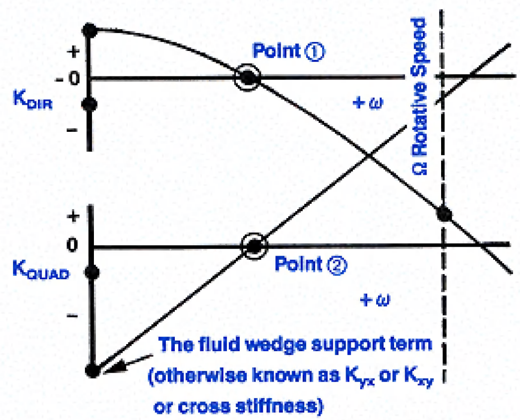

Figure 1

* *More correctly* $\omega = \dfrac{D_B\lambda\Omega}{D_B + D_S}$ *in the example and a little more complicated, but fundamentally the same, for typical rotor systems. See the papers of Donald E. Bently and Dr. Agnes Muszynska for details.*

These direct and quadrature stiffnesses of this very simple rotor apply quite well, no matter how complicated the actual rotor system. For any rotative speed Ω, the speed where the Direct Dynamic Stiffness K_{DIR} is zero is the well-known lateral resonance "critical" speed, point ① in Figure 1.

For that same rotative speed Ω, there is also a zero of the Quadrature Dynamic Stiffness terms K_{QUAD}. This is the fluid resonance, $\omega = \lambda\Omega$*, point ② in Figure 1. This fluid resonance is never seen except by perturbation methods like those described above.

When these two resonances occur at the same speed ω, then the rotor system whirls in accordance with well-known stability laws which govern the behavior of rotor systems as well as for any other closed loop control system.►

Since this full perturbation test has not been used in the specification for purchasing of machinery, here is a very simple test that can be done to give a vital portion of the results of the full test. This test is to use the weight of the rotor as a steady state ($\omega = 0$) perturbation force.

To accomplish this test, the gap voltages of the shaft observing XY probes at each end of the machine are observed while the machine is on the test stand in its normal position. Next, the machine under test is rotated to 45° left and the probe gaps are again observed. The machine is rotated 45° right and again the probe gaps are used. Of course, this has the effect of rotating the steady weight of the rotor in the bearings so that a gravity vector can be resolved. See Figure 2.

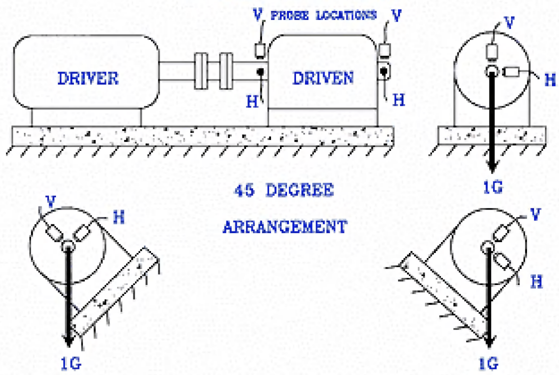

Figure 2

Call the measurement with rotation to the right A, and with rotation to the left B, and the weight of the rotor W. The "horizontal" force, F_H is $F_H = W \sin 45°$. See Figure 3.

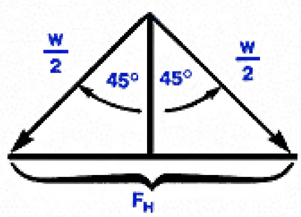

Figure 3

The motion vector change Z_H is $\overline{Z_H} = \overline{Z_A} - \overline{Z_B}$

The steady state ($\omega = 0$) stiffness vector for any rotative speed Ω is

$$K_{\omega=0} = \frac{F_H \angle 0°}{Z_H \angle A_{ZH}}$$

F_H is considered to be referenced at 0°. This $K_{\omega=0}$ is separated into two components:

$$K_{DIR\ \omega=0} = \left|\frac{F_H}{Z_H}\right| \cos A_{ZH} \text{ and}$$

$$K_{QUAD\ \omega=0} = \left|\frac{F}{Z_H}\right| \sin A_{ZH}$$

Now, throw away the direct term as it will usually contain the fluidic inertial effect term, which acts as a negative spring at $\omega = 0$ and confuses the results.

The remaining term is the steady state quadrature stiffness term. This term is $K_{QUAD\ \omega=0} = \lambda\Omega D$, the fluid support stiffness term. Plot this term on a Direct and Quadrature Stiffness graph, such as Figure 1. Add to this graph the line for rotative speed, Ω. Look up the self balance resonance, ω_{RES}, speed, and the "first critical" for this machine. Plot this on the zero stiffness line of the Direct Stiffness plot.

Now, look up the effective weight of the rotor, W_{EFF} (usually about 60% of actual rotor weight, the result of distributed weight between bearings, as compared to a concentrated load between bearings). Calculate the system effective direct stiffness $K_{DIR} = (\omega_{RES})^2 \frac{M_R}{2}$

Plot this K_{DIR} on the $\omega = 0$ line of the Direct Dynamic Stiffness Graph. You can now draw the negative parabola, $K_{DIR} - \omega^2 M_{EFF}$ on the Direct Dynamic Stiffness Graph.

As for the Quadrature Stiffness Graph, it is always a straight line with a slope, D, for damping since the entire line is $K_{QUAD} = (\omega - \lambda\Omega)D$. Rotative speed, Ω, is known, and you have already plotted K_{QUAD} at $\omega = 0$.

This leaves the λ ratio to be determined. With no antiswirling, a fair bet is a little less than ½, say 0.48. (Beware, some pumps and some high pressure compressors have λ ratios of anywhere from 0.5 to 1.2!) Plot this λD term on the $K_{QUAD=0}$ line of the Quadrature Stiffness Graph and draw a straight line from the $K_{QUAD\ \omega=0}$ fluid wedge support stiffness term through the $\lambda\Omega$ point. This is the Quadrature Stiffness Graph.

If the manufacturer has installed antiswirl to enhance stability, this data may be done with and without the antiswirl. This will exhibit and measure the corrective effect (reduction of the quadrature stiffness) of the stability control method.

The bottom line is that, for a stable machine, the λD term on the K_{QUAD} graph had better be SMALLER than the ω_{RES} point on the K_{DIR} graph! When they are at the same frequency, instability begins. If the λD term is larger, then ω_{RES} moves to a matching larger term merely by increasing the orbital size of the whip due to the instability. (A bigger size of orbit means a bigger bearing and seal stiffness, therefore, the higher "critical" speed.

It is obvious that this simple test takes some careful handling of alignment, so as not to introduce another steady term besides gravity, and requires the modal probes to reflect the motion from the probe locations to the centers of seals or bearings. It is also obvious that you can have the PREDICTION of stability or instability if that is important to you. ■